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ABOUT THE INFLUENCE OF THE CLEARANCE BETWEEN THE WORKING BLADES AND HOUSING OF A RADIAL TURBINE ON ITS EXPONENT

by

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FOREIGN TECHNOLOGY DIVISION

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09 English Title ABOUT THE INFLUENCE OF THE CLEARANCE BETWEEN THE WORKING BLADES AND HOUSING OF A RADIAL TURBINE ON ITS EXPONENT

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ABSTRACT The results of a 1959 experimental study of dissipation in an air radial turbine are reported. The relative internal turbine efficiency was determined; Re number variation was $(1.1 - 1.7) \times 10^6$; turbine pressure drop, 1.25 and 1.67; gap variation, 0.15 — 7 mm. It is found that the gap (clearance) loss plays an important role in the axial part of the rotor, and only a very little role in the radial part. The gap swirl results in a gap-to-channel gas flow, and this phenomenon reduces the gas leakage around the rotor-blade tips. The above phenomenon is also enhanced by Coriolis forces. In subsequent testing of various radial turbines, it was noticed that the turbine efficiency depended only very slightly on the gap value. It is found that: (1) The gap loss is considerably lower in radial centripetal turbines than in axial turbines; hence, the gaps in radial turbines need not be very small; (2) An empirical formula has been developed for calculating the gap loss; (3) A gas-dynamic scheme is suggested to explain the low gap loss. Orig. art. has: 7 figures and 14 formulas. English Translation: 13 pages.

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U. S. BOARD ON GEOGRAPHIC NAMES TRANSLITERATION SYSTEM

Block	Italic	Transliteration	Block	Italic	Transliteration
А а	*А а*	A, a	Р р	*Р р*	R, r
Б б	*Б б*	B, b	С с	*С с*	S, s
В в	*В в*	V, v	Т т	*Т т*	T, t
Г г	*Г г*	G, g	У у	*У у*	U, u
Д д	*Д д*	D, d	Ф ф	*Ф ф*	F, f
Е е	*Е е*	Ye, ye; E, e*	Х х	*Х х*	Kh, kh
Ж ж	*Ж ж*	Zh, zh	Ц ц	*Ц ц*	Ts, ts
З з	*З з*	Z, z	Ч ч	*Ч ч*	Ch, ch
И и	*И и*	I, i	Ш ш	*Ш ш*	Sh, sh
Й й	*Й й*	Y, y	Щ щ	*Щ щ*	Shch, shch
К к	*К к*	K, k	Ъ ъ	*Ъ ъ*	"
Л л	*Л л*	L, l	Ы ы	*Ы ы*	Y, y
М м	*М м*	M, m	Ь ь	*Ь ь*	'
Н н	*Н н*	N, n	Э э	*Э э*	E, e
О о	*О о*	O, o	Ю ю	*Ю ю*	Yu, yu
П п	*П п*	P, p	Я я	*Я я*	Ya, ya

* ye initially, after vowels, and after ъ, ь; e elsewhere. When written as ё in Russian, transliterate as yё or ё. The use of diacritical marks is preferred, but such marks may be omitted when expediency dictates.

FOLLOWING ARE THE CORRESPONDING RUSSIAN AND ENGLISH

DESIGNATIONS OF THE TRIGONOMETRIC FUNCTIONS

Russian	English
sin	sin
cos	cos
tg	tan
ctg	cot
sec	sec
cosec	csc
sh	sinh
ch	cosh
th	tanh
cth	coth
sch	sech
csch	csch
arc sin	$\sin^{-1}$
arc cos	$\cos^{-1}$
arc tg	$\tan^{-1}$
arc ctg	$\cot^{-1}$
arc sec	$\sec^{-1}$
arc cosec	$\csc^{-1}$
arc sh	$\sinh^{-1}$
arc ch	$\cosh^{-1}$
arc th	$\tanh^{-1}$
arc cth	$\coth^{-1}$
arc sch	sech^{-1}
arc csch	csch^{-1}
rot	curl
lg	log

ABOUT THE INFLUENCE OF THE CLEARANCE BETWEEN THE WORKING BLADES AND HOUSING OF A RADIAL TURBINE ON ITS EXPONENT

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Experience shows that during small expenditures of gas radial centripetal turbines have a higher eff than axial turbines. In those cases where the expenditure composes a tenth or hundredth part of kg/s (for instance, in turbine-driven compressors for supercharging diesels or either gas or pneumatic turbine drives) the fall of eff of axial turbines is so great that their use becomes either inexpedient or impossible. Radial turbines during such expenditures preserve an eff which is close to 80 or 85%. This circumstance deserves the most intent attention both from the practical and theoretical points of view.

Wide-spread affirmations [1] that the higher eff's of radial turbines are explained by the uniqueness of the fields of force in radial turbine channels ("Coriolis forces") have no real value, inasmuch as contemporary hydrodynamics does not yield a possibility of establishing the advantage of any force system from the point of

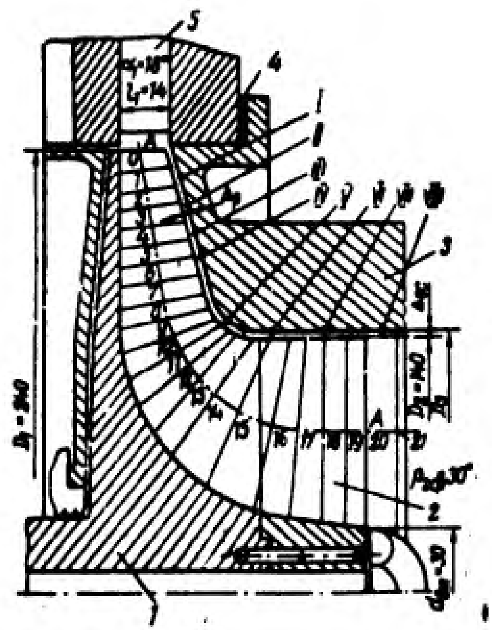

Fig. 1. Basic dimensions of a turbine: 1 — half-open rotor wheel; 2 — revolving guiding device; 3 — insert; 4 — packing; 5 — nozzle box assembly. I-VIII — points of measurement of static pressure.

view of the dissipative effects caused by them. It is necessary to experimentally reveal a concrete mechanism ensuring the essential distinction of dissipative effects in axial and radial turbines. Below it is shown that one of the important causes which explains the high eff of low-powered radial turbines is the negligible losses connected with magnitude of the clearance between the rotor wheel and housing and the losses, usually called "leakages."

In the article the results are presented for an experimental investigation carried out on an air radial turbine, the basic data for the blading of which are shown in Fig. 1. The investigation was conducted during the change of the complete clearance Δ on the whole and during the change separately of its radial Δ_p and axial Δ_{oc} parts. The change of the clearance Δ_p was produced by the shift of insert 3 in an axial direction and the clearance Δ_{oc} was changed by the boring of the insert along diameter D_3.

The total result of all the dissipative effects appearing with an increase of the clearance was evaluated by the change of relative internal eff of turbine $\Delta\eta_{01}$, where $\eta_{01} = 1 - \Sigma\xi_1$. Here $\Sigma\xi_1$ is the sum of relative losses in the nozzle box assembly, in the rotor wheel, with discharge velocity, from friction of the disk and with leakages.

Magnitude η_{0i} was determined by the drop of braking temperatures at the entrance and exit of the turbine.

It is known that the determination of eff of a radial turbine by temperature drops usually gives noticeable error. Therefore, for control we also determined the change of the effective eff of turbine $\Delta\eta_{0e}$ (by a hydraulic brake). In all cases we observed a good convergence of magnitudes $\Delta\eta_{0i}$ and $\Delta\eta_{0e}$ (see Fig. 2).

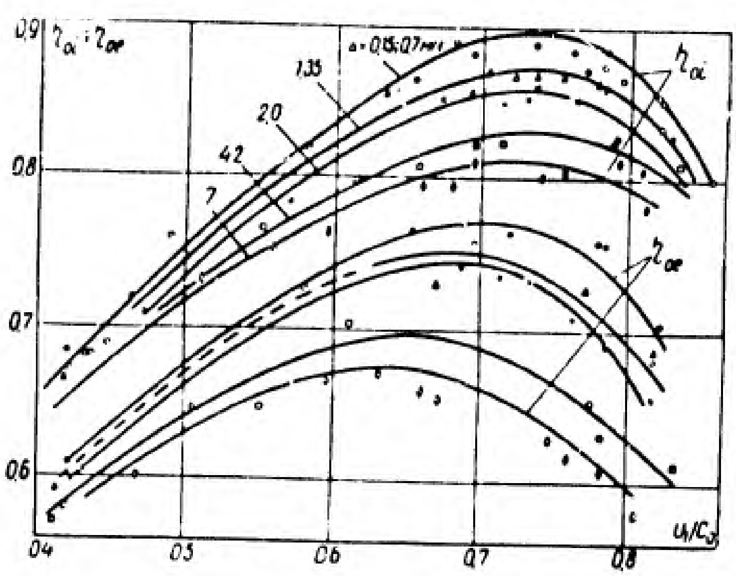

Fig. 2. Characteristic of a turbine in terms of eff at various clearances Δ, π = 1.67, ● – clearance, Δ = 0.15 mm; ○ – 0.70; △ – 1.35; × – 2.00; □ – 4.0; ◇ – 7.0.

The number $\mathrm{Re} = \frac{u_1 D_1}{2\nu_0}$ determined by viscosity of gas at the entrance of turbine ν_0, by the outside radius of the rotor wheel $D_1/2$, and by peripheral velocity u_1 was changed within the limits from $1.1 \cdot 10^6$ to $1.7 \cdot 10^6$. Experiments were conducted at two values of pressure drop on the turbine: $\pi = \frac{p_0}{p_2} = 1.25$ and $\pi = 1.67$; where p_0 is the total pressure before the nozzle box assembly and p_2 is the static pressure averaged by area in the output pipe of the turbine.

The clearance was changed from 0.15 mm (bundled insert [no exact definition of tal'kirovannaya (bundled) found]) to 7 mm (50% of the height of the blade at the entrance of the wheel). The results were processed in thr form of dependences:

$$\eta_{oi} = f_1\left(\frac{u_1}{C_0}, \pi\right);$$
$$\frac{G\sqrt{T_0}}{p_0} = f_2\left(\frac{u_1}{C_0}, \pi\right);$$
$$\rho = f_3\left(\frac{u_1}{C_0}, \pi\right),$$

where G — expenditure of gas, kg/s; C_0 — distributed isoentropic speed, m/s; ρ — degree of reactance in terms of adiabatic thermodrops.

<u>Results of experiments.</u> Influence of the magnitude of clearance Δ between the wheel and housing of a turbine on its internal eff is shown in Fig. 2.

With the increase of clearance Δ from 0.15 to 1 mm $\left(\frac{\Delta}{l_1} = 1\text{-}7\%\right)$ the change of eff η_{01} does not go beyond the limits of accuracy of the experiment. With increase of gap Δ to 7 mm $\left(\frac{\Delta}{l_1} = 50\%\right)$ the eff

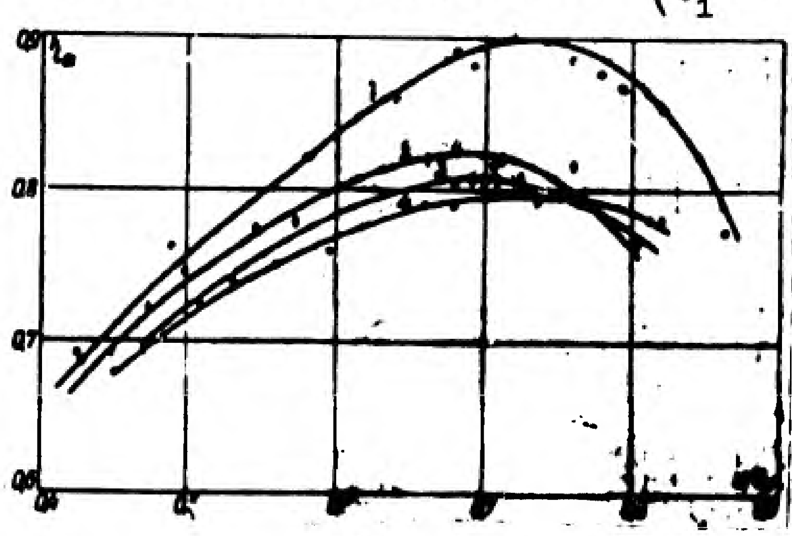

Fig. 3. Characteristic of a turbine by its relative eff at the biggest and least clearances Δ_p and Δ_{oc}. 1 — Δ_p = 0.7 mm; Δ_{oc} = 0.7 mm; 2 — Δ_p = 7.0; Δ_{oc} = 0.7; 3 — Δ_p = 0.7; Δ_{oc} = 7.0; 4 — Δ_p = 7.0, Δ_{oc} = 7.0.

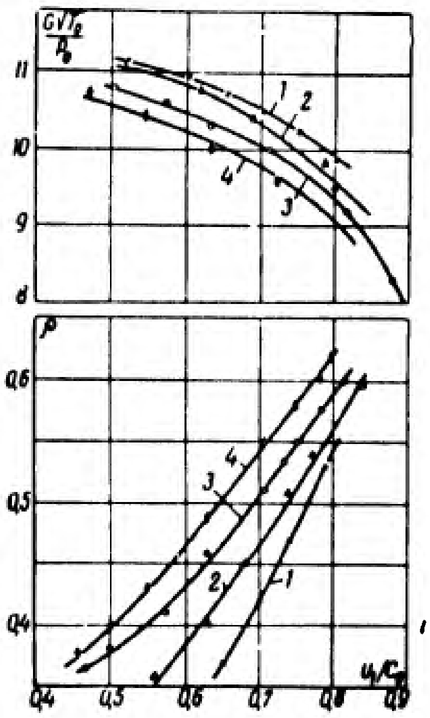

Fig. 4. Characteristics of a turbine by the degree of reactance and expenditure at various values of clearances Δ_p and Δ_{oc}. $\pi = 1.67$, 1 – $\Delta_{oc} = 7$ mm, $\Delta_p = 0.7$ mm; 2 – $\Delta_{oc} = \Delta_p = 7$; 3 – $\Delta_{oc} = 0.7$, $\Delta_p = 0.7$; 4 – $\Delta_{oc} = 0.7$, $\Delta_p = 7$.

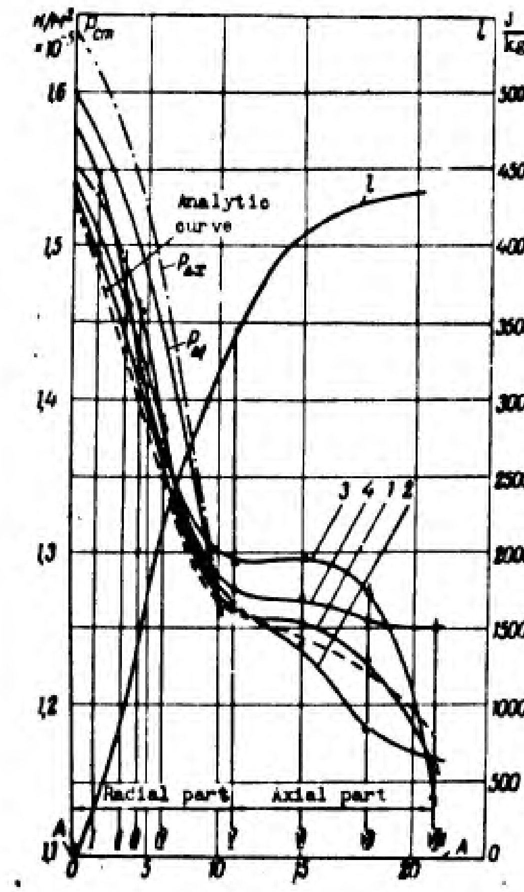

Fig. 5. Distribution of static pressures in the clearance between the wheel and housing of a turbine $\pi = 1.67$; $u_1/C_0 = 0.735$; 1 – $\Delta_{oc} = \Delta_p = 0.7$ mm; 2 – $\Delta_{oc} = 0.7$ mm, $\Delta_p = 7$ mm; 3 – $\Delta_{oc} = 7$ mm, $\Delta_p = 0.7$ mm; 4 – $\Delta_{oc} = \Delta_p = 7$ mm. l – distribution of work of gas by length of channel.

at the design point drops 9.5%. At off-design points $\frac{u_1}{C_0} \gtrless \left(\frac{u_1}{C_0}\right)_{опт}$ the influence of the clearance is less than at the design point.

Figure 3 shows the results of the determination of η_{0i} during the separate change of the axial Δ_{oc} and radial Δ_p parts of the clearance. For simplification of the graphs curves are given only for the following combinations of clearances: $\Delta_{oc} = \Delta_p = 0.7$ mm; $\Delta_{oc} = 0.7$ mm, $\Delta_p = 7$ mm; $\Delta_{oc} = 7$ mm; $\Delta_p = 0.7$ mm; $\Delta_{oc} = \Delta_p = 7$ mm.

Figure 4 shows the corresponding dependences for the given expenditure and degree of reactance.

Finally, Fig. 5 shows the distribution of static pressure along the inner surface of the housing of a turbine at a design point ($u_1/C_0 = 0.7$) for the same four combinations of clearances and the value of work l, which is transmitted from the gas to the wheel.

The work was calculated by the change of average braking enthalpy

$$l = c_p(T_1 - T_2) + \frac{c_1^2 - c_2^2}{2g}. \qquad (1)$$

Calculations show that in a given turbine at small values of clearance Δ approximately 78% of the work is yielded by the gas in the radial part of the rotor wheel (from the entrance section to section No. 11) and only 22% in its axial part.

Along the axis of abscissas, in Fig. 5 the expanded axis of channel A-A is given. The experimental points correspond to pressures measured through holes at points I-VIII (see Fig. 1).

The data given in Fig. 3 shows that the increase of only one axial part of the clearance Δ_{oc} causes essentially larger losses than a change of its radial part Δ_p. If one were to consider that the gas approaches the axial part of the clearance preserving in all 22% of the available thermodrop, then carrying the loss from growth of this part of the clearance to the indicated portion of the thermodrop we will obtain loss coefficient $\zeta'_{\Delta oc} = \zeta_{\Delta oc}\frac{1}{0.22}$, i.e., the relative growth of losses is almost 5 times. The growth of only one radial part of clearance Δ_p has a much smaller influence on the eff of a turbine.

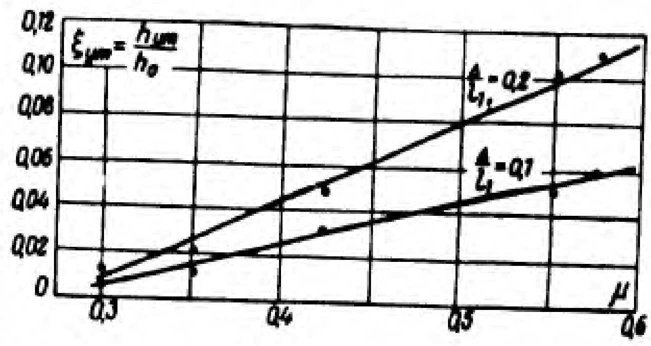

Fig. 6. $\xi_{ут} = \frac{2\Delta}{l_1}(\mu - 0{,}275)$

The data given in Fig. 4 show that increase of only one axial part of clearance Δ_{oc} leads to an insignificant growth of expenditure. If the whole clearance Δ is increased the growth of expenditure turns out to be smaller and the increase of only the radial part of clearance Δ_p leads not to an increase but to a decrease of gas expenditure.

Thus experimental data distinctly show that losses "in leakages," which play a considerable role in the axial part of the wheel, are minute in its radial part. Hence it directly follows that losses in "leakages" depend on the degree of radiality of the rotor wheel $\mu = \frac{D_{1cp}}{D_1}$ (see Fig. 1). The greater the value μ, then the higher the losses in leakages have to be at the same Δ/l_1.

Figure 6 shows the dependence $\xi_{ут} = f(\mu)$ for two values of the clearance Δ/l_1 obtained by the data of the experiments. This dependence is well approximated by the equation

$$\xi_{ут} = \frac{2\Delta}{l_1}(\mu - 0{,}275), \tag{2}$$

which it is possible to recommend for an appraisal of losses through "leakages" in radial turbines. It is clear that this equation, as does any purely empirical dependence, requires additional experimental refining. Let us try to show how small losses through "leakages" are explained in radial turbines with a small value of μ.

From comparison of curves of static pressure in Fig. 5 one can see that an increase of only one axial part of the clearance does not cause an essential change of static pressure along the flow part of the turbine. Noticeable growth of pressure in the beginning of the axial section is explained by geometric influence (increase of flow area). With increase of the whole clearance there is observed an essential growth of static pressure on the periphery of the axial part of the operating canal which attests to the appearance of spin in the peripheral part of the flow. Increase of only one radial part of the clearance leads to a significant fall of static pressure in the exit part of the channel.

Let us try to compose a gas-dynamic pattern which explains these results.

The data shown in Fig. 4 and 5 (decrease of expenditure on account of growth of clearance Δ_p and increase of static pressure at the exit from the wheel) clearly show that in the radial part of the clearance there exists a turbulent movement caused by a spin of air by the nozzle box assembly of the turbine and leakage of gas through the clearance.

Let us consider how the parameters of gas in the clearance are changed if the clearance is isolated from the operating canals.

In the simplest case of motion without friction, we have, due to the absence of a moment of external forces, an evident condition for the peripheral component of speed: $c_u R = \text{const}$. Using the equation of energy for motion of gas in a clearance and considering $c_{ux} = \frac{R_1}{R_x} c_{u1} = k_x c_{u1}$, we will obtain a value of the expended component of speed in any section of the clearance

$$c_{rx}^2 = c_{r1}^2 - c_{u1}^2 (k_x^2 - 1) + 2c_p T_{\Delta 1}\left(1 - \frac{1}{\pi_{\Delta x}^{\frac{k-1}{k}}}\right). \qquad (3)$$

where $\pi_{\Delta x} = \frac{p_1}{p_{\Delta x}}$ – the pressure drop in the isolated clearance; and $p_{\Delta 1}$, $T_{\Delta 1}$, $c_{\Delta 1}$ – the parameters of gas at the entrance to the clearance.

From the equation of expenditure, considering that the area of the clearance is $F_{\Delta x} = 2\pi R_x \cdot A$, we will obtain

$$c_{rx} = \frac{R_1 \gamma_1}{R_x \gamma_x} c_1 \sin \alpha_1 = k_x \frac{\gamma_1}{\gamma_x} c_1 \sin \alpha_1. \quad (4)$$

The change of density we will define by isoentropy

$$\gamma_x = \gamma_0 \left(\frac{1}{\pi_c} \pi_{\Delta x} \right)^{\frac{1}{k}} = \gamma_0 \varepsilon, \quad (5)$$

where $\pi_c = \frac{p_0}{p_{\Delta 1}}$ – the pressure drop in the nozzle box assembly; ε – gas-dynamic density function.

With help of equations (3)-(5) one can find the parameters of gas in any section of the isolated clearance if boundary conditions are assigned.

Let us say that at the entrance to the axle part of the operating canal (section 11 in Fig. 5) there is a static pressure in the clearance equal to the average static pressure in this part of the channel.

Taking condition $p_{\Delta 11} = p_{11}$, we obtain for the isolated clearance a distribution of pressure along curve $p_{\Delta x}$ (Fig. 5). During this the parameters of gas at the entrance to the clearance are not equal to parameters at the entrance to the wheel. In particular, the pressure $p_{\Delta 1} > p_1$ which was even confirmed by the experiment (decrease of expenditure) with the growth of clearance Δ_p.

The calculation of distribution of pressure in the clearance can be refined if one considers the influence of friction forces. Let us use for the given case the pattern of the calculation of pressure in the clearance between the wheel and housing of the centrifugal pumps offered by A. A. Lomakin [5] and considered by

L. A. Dorfman [6]. The corresponding distribution of pressure in an isolated clearance for the same boundary conditions as in the case without friction is shown in Fig. 5 by curve $p_{\Delta r}$. Comparison of curves $p_{\Delta x}$ and $p_{\Delta r}$ shows that the influence of friction forces does not change the qualitative conclusions obtained on the basis of isoentropic pattern.

The presence of spin in the clearance leads to an verflow of gas from the clearance to the channel and these overflows decrease the leakage of gas through the edges of the blades of the rotor wheel, i.e., they remove the most important cause of losses "with leakages."

Let us note two more circumstances which promote decrease of leakages through the clearance in a radial turbine. The overflow of gas through the edges of blades is impeded not only by the spin in the clearance but also by Coriolis forces which influence the overflowing stream of gas so that they displace it along the edge of the rotor wheel to the periphery, i.e., from the region of the smaller pressures of the channel to the region of the larger (Fig. 7a).

a) b)

Fig. 7.

Besides in the channel of an axial turbine the increase of the open radial clearance has a two-fold action on losses with leakage: besides growth of direct overflow through the clearance there occurs a strengthening of radial flows from the concave wall of a blade to the periphery and from the convex to the center [7]. The influence of Coriolis forces on these flows leads to the intensifying harmful innerchannel spin in the upper part of the channel (see Fig. 7b).

In the channel of a radial turbine on the Coriolis forces do not act on the stream flowing from the blade root to the periphery, and strengthening of innerchannel spin does not occur.

Thus both the quantity of gas overflowing through the edges of blades and the consequences of such overflowing for radial turbines are considerably less than for axial turbines.

The weak influence of clearance Δ_p on the eff of a radial turbine can be used for the design of blading of turbines with low heights of nozzle blades and also for purposes of adjustment of turbines working in the narrow range of u_1/C_0.

The work was carried out in 1959. For the past four years we have tested several radial turbines of different designation. In all cases we noticed a very weak dependence of their eff on the values of clearance Δ_p.

In conclusion we will note a curious experimental result.

The relative speed in the channel of the rotor wheel at a radius R_x

$$w_x = \psi \sqrt{w_1^2 - u_1^2 + \mu_x^2 u_1^2 + 2\frac{k}{k-1} RT_1 \left[1 - \frac{1}{\pi^{\frac{k-1}{k}}}\right]}. \tag{6}$$

where

$$\mu_x = \frac{R_x}{R_1};\ \pi_x = \frac{p_1}{p_x}.$$

The continuity equation gives

$$w_1\gamma_1 F_1 = w_x \gamma_x F_x, \tag{7}$$

where F_1, F_x – the areas of the operating canal at the entrance to the wheel and in the section at a radius R_x. And, finally, the politrope equation

$$\frac{\gamma_1}{\gamma_x} = \left(\frac{p_1}{p_x}\right)^{\frac{1}{n}}, \tag{8}$$

where

$$n = \frac{k}{\psi^2 + k(1-\psi^2)}. \tag{9}$$

From equations (6)-(9) it is possible at the assigned value of ψ and the degree of reactance ρ to find the static pressure p_x.

For comparison of calculated and experimental data in Fig. 5 to the calculated values of p_x in the axial part of the channel we added the value $\Delta p = \gamma \frac{u_R^2 - u_m^2}{2}$, where u_R and u_m – peripheral velocities of the axial part of the channel and at its center line.

The experimentally found distribution of static pressure along the operating canal with a small clearance (curve 1 in Fig. 5) coincides well with the calculated distribution of this pressure obtained on the basis of a one-dimensional model (dotted line in Fig. 5).

Conclusions

1. It is experimentally shown that losses in leakages in radial centripetal turbines at small values of μ are essentially less than in axial turbines. This circumstance should be considered with the designation of technological clearances between the wheel and housing which it is not necessary to make small for radial turbines and also with a comparative appraisal of radial and axial turbines.

2. On the basis of experimental data on many radial turbines we obtained an empirical formula for calculation of losses in leakages.

3. We offered a gas-dynamic pattern which explains the lowering of losses from overflows by spin motion in the clearance and by the influence of Coriolis forces on the overflowing streams of gas.

4. We noticed the possibility of using the small sensitivity of the eff of radial turbines with a small value of μ to the value of the clearance for purposes of adjustment.

The conclusions obtained have an especially important meaning for calculation and designing of radial turbines of turbosuperchargers of low power, gas-turbine engines, and starting turbines.

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